HEAT TRANSFER IN AN EIGHTY SQUARE FEET SURFACE CONDENSER

by

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INTRODUCTION

Heat transfer from condensing steam to water flowing inside a tube is usually studied by constructing a single tube condenser. This method has some advantages, but it also has several very serious disadvantages. There is no doubt that it is possible to get more complete data on the performance of a single tube condenser, but it is doubtful if the results obtained from a single tube can be applied to condensers having many tubes without modification.

The power plant condenser contains many hundreds of these tubes staggered and placed very close together. The problems that cannot be evaluated by means of a single tube condenser are (1) the distribution of steam between the various tubes, (2) the distribution of the water between the tubes, and (3) the concentration of air-vapor mixture around the tubes.

In the design of surface condensers, it is generally assumed that the same amount of water goes through all the condenser tubes. This is not likely to be true. Therefore, it was felt that the distribution of water in a surface condenser would be important data. The velocities of the water in the 42 tubes were measured without disturbing the flow pattern any more than was necessary. The velocity of the water through the tubes affects the water film coefficient, and thereby the over-all coefficient.

However, the individual tube heat transfer data were considered the most important. The heat transferred in each of the tubes of the lower pass was determined. These data may be used to check equations now in use, and to check the merit of the way the tubes are placed in the condenser.

MATERIALS AND METHODS

A Worthington two-pass eighty square feet surface condenser was used. There were 42 tubes in the lower pass, and 40 tubes in the upper pass, each tube being 5/8 inch outside diameter and 18 gauge. These tubes were fastened into a tube sheet at each end of the condenser, and made water tight by means of ferrules.

At the downstream end of the lower pass were constructed pitot tubes facing into the water stream by means of which the velocities in the condenser tubes were determined. These pitot tubes were made of copper tubing .040 inch inside diameter, and were not made of standard design because of the difficulties encountered. However, by calibration it was possible to determine the true velocities. The readings obtained were pressure differences between the static pressure and the total pressure. These readings were equivalent to $\frac{Kv^2}{2g}$ where K = calibration constant, v = water velocity in feet per second, and g = acceleration of gravity = 32.2 feet per second per second. The deflection of a manometer for a velocity of 2 feet per second would be about 3/4 inch of water. However, a two fluid manometer was used. The fluid other than water was nitrobenzene which has a density of 1.198. This gave a multiplication factor of 5.05, so that for a velocity of 2 feet per second, the deflection was 3.75 inches. It is obvious that the reading accuracy was much improved.

It would have been awkward to have 42 manometers so that all readings could be taken simultaneously. Therefore, a method of reading the deflections for all 42 tubes in rapid order on one manometer was devised. A diagram of this equipment appears in Plate II. For any position of the sliding piston,

only one pitot tube is connected to the manometer. The connection at the top measures the total pressure, and the connection at the bottom measures the static pressure of the pitot tube in use. This condenser was placed so that the tubes were running east and west. The numbers of condenser tubes are shown in Plate III, and it should be noted that the north half and the south half have like numbers. The nomenclature used, therefore, is 1N, 1S, etc. to represent number 1 on the north side, number 1 on the south side, etc. The construction used eliminated roving pitot tubes. Roving pitot tubes were tried, but they were frequently damaged in their movements about the condenser. To change from one tube to another, it was necessary only to turn the hand screw to the position desired as indicated by the pointer. One thing that gave trouble was the collection of air in the rubber tubes leading to the manometer. When this occurred, an error in the readings appeared. This difficulty was minimized by careful bleeding of the rubber tubes to the manometer, and by accepting only such readings as checked on two separate days.

The calibration of the pitot tubes was determined by forcing a known amount of water through each of the condenser tubes in order. One tube was separated by removing the ferrule on the inlet end, and inserting a ferrule which has attached a nipple. A small orifice meter was constructed, and the outlet connected by means of a rubber tube to the above mentioned nipple. This orifice was calibrated by weighing the water and noting the deflection of the mercury manometer. These data were plotted (Fig. 1). After the calibration was complete, the outlet water from the orifice was forced through one of the condenser tubes, and the nitrobenzene manometer deflection for this condenser tube noted. This was repeated for several tubes. Since all pitot tubes were identical, and because all readings taken checked within 3 percent,

it was decided that the average of the calibration data obtained on six tubes could be used for all the pitot tubes.

Thermocouples were placed near the pitot tubes, and in the end of the condenser tubes. In order that they read only the water temperature, there was no direct contact with the metal of the condenser tubes. The purpose of these thermocouples was that of giving the final water temperature so that the temperature rise could be found. The product of the weight of water and the temperature rise gives the heat transferred.

In order to calibrate these thermocouples, it was decided that one temperature would be checked in the condenser. Then the remaining calibration was conducted on a thermocouple made of the same wire as the thermocouples in the condenser.

The steam used in this experiment was taken from the 100 p.s.i. College steam line. This steam, of course, was reduced in pressure and desuperheated until only 10 or 15 degrees of superheat remained. The desuperheating was accomplished by injecting water into the steam.

This condenser was operated at atmospheric pressure. This means that the total pressure through the condenser remained at atmospheric pressure. Dalton's law states that the sum of the partial pressures must be equal to the total pressure. Or, applied to this problem, it means that at any point in the condenser, the sum of the air and steam pressures must be equal to atmospheric pressure. Because of the fact that no appreciable air was added with the steam, no air came from the condenser after equilibrium was established. However, for any particular point in the condenser, the percentage of steam and air changed when the amount of entering steam was changed. This was, of course, reflected in the temperature readings. The temperature of the inlet

steam, the temperature of the steam between passes, and the steam temperature at the bottom of the last pass were taken.

The condensate was removed from the condenser by means of a centrifugal pump. This pump forced the condensate into a barrel where it could be weighed if desired. This was found to be not necessary because it could not serve as a good check. Only one half the condenser was being used to obtain data.

Finally, a check was made of the velocity profiles at various Reynolds numbers. Scale, which causes a certain amount of roughness, will sometimes change the velocity profile. The curves appear in Fig. 5.

REVIEW OF IMPORTANT LITERATURE

Heat transfer is usually studied by constructing a single-tube condenser. Such a procedure is much easier than that used by the writer. No literature giving a full detailed analysis of the heat transferred in a multi-tube condenser was available. Therefore, the majority of the literature reviewed must apply to single-tube condensers only. Certain factors used in equations for single tube condensers cannot be evaluated accurately for the multi-tube condenser.

Colburn and Hougen (1930) have probably given the best study of the theory of heat transmission. They stated that the film coefficient for forced convection is k/d times a function of the Reynolds number, the Peclets number, the Stantons number, and L'. A list of heat transfer symbols follow:

$Re = \frac{Vd}{\mu} = \frac{\rho ud}{\mu}$ (Reynolds number)

$Pe = \frac{k}{Vdc_p} = \frac{k}{u\rho dc_p}$ (Peclets number)

$Gr = \frac{d^3 \rho^2 \; g \Delta t \; \beta}{\mu^2}$ (Grashofs number)

$St = \frac{k}{\mu \, c_p}$ (Stantons number)

$L' = \frac{L}{d}$

V = mass velocity of fluid

u = linear velocity of fluid

ρ = density of fluid

d = linear dimension of length or diameter

μ = absolute viscosity of fluid

k = thermal conductivity of fluid

c_p = heat capacity of fluid (specific heat at constant pressure)

g = acceleration of gravity = 32.2 ft. sec.$^{-2}$

Δt = temperature drop across film

β = coefficient of thermal expansion of fluid

L = length of heat exchanger

The Reynolds number, Peclets number, Grashofs number, and Stantons number are all dimensionless quantities. The advantages of dimensionless numbers are that they may be raised to any power without disturbing the units of the remaining parts of the equation, and that any consistent system of units may be used without disturbing the equation or any of the constants.

Rice (1927) used the following general relation for all fluids:

$$h_g = \frac{k}{d} \text{ constant } \left(\frac{\mu \, c_p}{k}\right)^a \left(\frac{Vd}{\mu}\right)^b \qquad (1)$$

h_g = film coefficient

By means of this general expression, heat transfer data may be correlated.

However, the problem of finding the conductivity of the air-steam film is

somewhat more difficult. The two factors involved in this problem are (1) the heat transferred by the diffusion of the steam to the colder condensate film, and (2) the mass flow of the mixture towards the condensate film. The total amount of steam reaching the condensate film is the sum of the amounts transferred by the above two methods. The amount of vapor moving by diffusion is:

$$D_v = -K_d \frac{\delta \rho_v}{\delta x} \tag{2}$$

D_v = the rate of diffusion in mass per unit area, and unit time

K_d = the diffusion coefficient, mass per unit time, unit area, unit concentration gradient

ρ_v = the density (or concentration) of the vapor, mass per unit volume

x = distance through which diffusion is taking place

By the gas law, $\rho_v = \frac{M_v P_v}{R\,T}$ where

M_v = the molecular weight of the vapor

P_v = the partial pressure of the vapor

R = the gas constant

T = the absolute temperature

Therefore, $D_v = -\frac{M_v K_d \delta P}{R\,T \delta x}$ (3)

Boynton and Brattain (1929) state that the diffusion coefficient for a vapor into a gas is the same as the diffusion coefficient for the gas into the vapor. Because the pressure gradient of the gas is the same as that of the vapor except for sign, the gas will also diffuse according to the same relation.

$$D_g = -\frac{M_g K_d \delta P_g}{R\,T\,\delta x} \tag{4}$$

$$\frac{\delta P_g}{\delta x} = -\frac{\delta P_v}{\delta x}$$

$$D_g = \frac{M_g K_d \delta P_v}{R\,T\,\delta x} \tag{5}$$

For the case of steady state vapor diffusion, the total and partial pressures of the gas and vapor remain constant at any point. For this condition to be maintained, it is necessary that a sufficient amount of the mixture flows toward the condensing surface so that the gas in this mixture is just equal to the amount of gas diffusing away from this surface. Therefore:

$$F_g = - D_g = - \frac{M_g K_d \delta P_v}{R\ T\ \delta\ x} \tag{6}$$

F_g = the rate of flow of the gas in mass per unit area

$$P_v V_v = F_v \frac{R}{M_v} T \quad (R = 1544) \tag{7a}$$

$$P_g V_g = F_g \frac{R}{M_g} T \tag{7b}$$

$$V_v = V_g$$

$$F_v = F_g \frac{M_v P_v}{M_g P_g} \tag{8}$$

$$F_v = - \frac{M_v K_d \delta P_v\ P_v}{R\ T\ \delta\ x\ P_g} \tag{9}$$

The total vapor transfer is the sum of D_v and F_v.

$$D_v + F_v = - \frac{M_v K_v}{R\ T} \left(\frac{\delta P_v}{\delta\ x} \right) \left(1 + \frac{P_v}{P_g} \right) \tag{10}$$

$$(D_v + F_v) \frac{R\ T}{M_v K_d} dx = \left(1 + \frac{P_v}{P - P_v}\right) (- dP_v) \tag{11a}$$

$$= \left(\frac{P}{P - P_v}\right) (- dP_v) \tag{11b}$$

where P = total pressure

However, K_d varies about as $T^{3/2}$, $\frac{K_d}{T}$ varies as only $T^{1/2}$, and may be assumed constant over a narrow temperature range. Integrating the above expression, the following equation is obtained:

$$D_v + F_v = \frac{M_v P\ K_d}{R\ T\ \phi} \log_e \frac{P_{g1}}{P_{g2}} \tag{12}$$

where ϕ is the film thickness

If the mass transmission coefficient, K, be defined as the mass of vapor transferred per unit area, unit time, and unit partial pressure difference; then the following equation applies:

$$K = \frac{M_v K_d}{R\ T\ \phi} \quad \frac{P}{(\log \text{ mean } P_g)} \tag{13}$$

Tinker (1933) modified the above equation and substituted experimental data to give the following equation:

$$h = c\ (T \text{ corr.})\ (D \text{ corr.})\ G^{.69} \frac{P_{vf}}{P_{gf}}\ f \tag{14}$$

h = air-steam film coefficient, Btu/hr-ft^2-$^\circ$F

$T \text{ corr.} = \left(\frac{560}{T}\right)^{1.8}$

T = degrees absolute (Rankine)

$D \text{ corr.} = \left(\frac{1}{d}\right)^{.31}$

d = tube diameter, inches

G = mass velocity, lb/hr-ft^2

P_{vf} = mean vapor pressure, in. Hg.

P_{gf} = log mean gas pressure, in. Hg.

f = film thickness factor depending on relative concentrations of gas and vapor or on $\frac{P_{vf}}{P_{gf}}$

c = a constant (given as 5 by Tinker)

$$P_{gf} = \frac{P_{gt} - P_{gm}}{\log_e \frac{P_{gt}}{P_{gm}}}$$

$P_{vf} = P - P_{gf}$

P_{gt} = P - (water vapor pressure corresponding to T_t)

P_{gm} = P - (water vapor pressure corresponding to T_m)

T_t = tube wall temperature, deg. F

T_m = air-vapor mixture temperature, deg. F

P = total mixture pressure, in. Hg.

Colburn and Hougen (1934) recommended a graphical solution of heat transfer problems involving dehumidification. Problems of this sort are not easily solved by ordinary means.

The various resistances are found first, and for any problem, certain of these resistances are constant. That is, the dirt, tube, water, and condensate resistances are added together to find a total fixed resistance. The reciprocal of this resistance gives the conductance of the combined films, henceforth to be known as h_o.

There are two means of getting heat transferred to the condensate film, and thereby eventually to the cooling water. These methods are by diffusion and by mass flow. The heat transferred by mass flow to the film and conduction across the film is given by the following equation:

$$h_s\ (t_g - t_c) \tag{15}$$

$$h_s = j\ c_pG/(\frac{c_p\mu}{k})^{2/3} \tag{16}$$

t_g = the gas temperature

t_c = the condensate temperature

c_p = the specific heat at constant pressure

G = the mass velocity

μ = the viscosity

k = the conductivity

j = heat transfer or mass factor which is a function of the Reynolds number (see Colburn, 1933)

In order to prove that the above equation is correct, it is necessary to prove that "j" can be expressed as a function of the Reynolds number. To do this, it is well to start with the general equation of Rice (Equation 1).

$$h = \text{const.}\ \frac{k}{d}\left(\frac{\mu c_p}{k}\right)^a \left(\frac{dG}{\mu}\right)^b \tag{17}$$

$$h = j\ c_p\ G/\left(\frac{\mu c_p}{k}\right)^{2/3} \tag{18}$$

Equating (17) and (18)

$$j = \text{const.}\ \left(\frac{\mu c_p}{k}\right)^{2/3} \frac{k}{c_p G d} \left(\frac{dG}{\mu}\right)^b \left(\frac{\mu c_p}{k}\right)^a \tag{19}$$

$$j = \text{const.}\ \left(\frac{\mu c_p}{k}\right) \frac{k}{c_p G d} \left(\frac{dG}{\mu}\right)^b \left(\frac{\mu c_p}{k}\right)^{a-1/3}$$

$$j = \text{const.}\ \left(\frac{dG}{\mu}\right)^{b-1} \left(\frac{\mu c_p}{k}\right)^{a-1/3} \tag{20}$$

According to kinetic theory, $\left(\frac{\mu c_p}{k}\right)$ is a constant.

$$j = \text{const.}\ \left(\frac{dG}{\mu}\right)^{b-1} \tag{21}$$

In addition to the heat transferred as calculated by Equation 15, there is also heat transferred by diffusion. The amount of heat transferred in this manner may be evaluated by the following equation:

$$KM_v \lambda (p_v - p_o) \tag{22}$$

$$K = \frac{j\ G}{M_m p_{gf} \left(\frac{\mu}{\rho K_d}\right)^{2/3}}$$

j = mass factor as previously used

M_m = average molecular weight of mixture

p_{gf} = pressure of gas in film

ρ = density

K_d = diffusion coefficient

$M_v\lambda$ = molar latent heat

p_v = pressure of vapor

p_o = pressure of saturated vapor at temperature t_o

Therefore, the heat transferred per square foot is given by the following equation:

$$h_s (t_g - t_o) + KM_v \lambda (p_v - p_o) = h_o (t_o - t_w) = U\ \Delta t \tag{23}$$

t_w = water temperature

The above equation may be solved by trial and error for various assumed points in the process of dehumidification. After this is completed, a curve of $10^6/U$ may be plotted against B.t.u. x 10^{-6}. From this curve, a new curve may be determined by integrating the area under the first curve. The last curve is a plot of area or surface required against heat transferred, and is the desired answer.

Robinson (1920, p. 644) calculated the heat transfer coefficient of the combined vapor and condensate film from the data of Kerr using an empirical equation for the water film coefficient. The following equation was derived for this air-steam coefficient:

$$\log h_v = 1 + 0.025\ Y \tag{24}$$

Y = volume percentage of steam

The above equation was plotted (Fig. 2). The percentage of air is usually given as a percent by weight. To aid in the changing over from a weight to volume percentage, the curves of Fig. 3 and Fig. 4 are presented.

Tinker (1933) showed that the use of a "cleanliness factor" is not desirable. The same tube operating under different conditions has different "cleanliness factors." Therefore, such a factor is not reliable. Instead, if a dirt coefficient is used, the performance of a tube is easily predicted. If dirt and scale would not accumulate, it would be possible to construct condensers with fewer tubes and thereby reduce the cost.

Tinker stated that the pressure drop in a tube bank is very hard to calculate. He gave the following equation for the pressure loss in a section of a tube bank:

$$\Delta P = e_1 L_1\ u^2\ P$$

ΔP = pressure loss, in. Hg.

c_1 = a constant depending on the tube configuration

P = absolute pressure, in. Hg.

u = mean vapor velocity through tube bank, ft/sec at the most restricted area

L_1 = vapor travel along curved path, ft.

If the tubes in the bank have a uniform square or diamond pitch arrangement, the following equation may be used:

$$\Delta P = c\ L\ u^2\ P/(S - D)$$

c = constant for coordinating units

L = straight line distance through tube bank, ft.

S = tube spacing center to center, in.

D = tube outside diameter, in.

This equation is an approximation based on the Fanning's equation, and the treatment of the tube bank as a completely rough conduit. The pressure drop is important because it affects the mean temperature difference.

An analysis of the condensation occurring in a surface condenser shows that the steam striking the cooler end of the tube condenses more rapidly than that striking the warmer end. This effect causes a flow along the tube, and hence, close fitting vertical baffles or tube supports should not be used. This condition is not so serious for the two pass design when the over-all performance is considered.

After a large percentage of the steam has been condensed, the temperature of the mixture begins to drop more rapidly. The author calls this section of the condenser the "devaporizing" section instead of the more commonly used term "air cooling" section. The more completely the air is devaporized, the smaller the ejector required. This devaporizing section is the most difficult

part of the condenser to analyze.

For the same design, more tubes in the devaporizing section mean a larger amount of the vapor condensed. This increases the cost of the condenser, but decreases the cost of the ejectors and increases the returned condensate if direct acting air pumps are used. In a practical application, a balance between these factors is necessary to give the best design from an economic viewpoint.

THEORY OF HEAT TRANSFER

One method of explaining the phenomena of heat transfer is that of the resistance concept. It is then possible to calculate the heat flowing if the temperature difference and resistance to heat flow are known in an identical manner, as it is possible to calculate the current flowing in an electrical circuit if the voltage drop and resistance are known. The total resistance is found by adding up the individual resistances.

In a steam condenser, the resistances to the flow of heat are (1) the water film, (2) the water side scale, (3) the metal wall of the tube, (4) the steam side scale, (5) the condensate layer, and (6) the air-vapor film.

The water in most condenser tubes flows in a turbulent manner. This means that the water follows a random motion. However, along the sides of the tubes, there is a layer of water flowing with viscous motion. This means that the water of this layer is flowing along in the direction of the axis of the tube. The velocity of this layer at the tube wall is zero, and increases in a straight line relation until the turbulent core is reached. The resistance of this outer layer to the flow of heat has been rather carefully worked out. In the range of condenser temperatures, the water film conductivity may be determined by the following empirical equation:

$$h_w = \frac{(154 + 1.6t)\ u^{.8}}{d.^2}$$

h_w = conductivity of water film

t = average water temperature, deg. F

u = average water velocity, ft/sec

d = diameter of the tube, inches

The water side scale and the steam side scale are very hard to determine. About the best way to determine the scale resistances is that of comparing the results obtained from the dirty tube with those obtained from a clean tube. However, for this experimental work, the combined resistance of the water side scale, the steam side scale, and the condensate film was used.

The resistance of the tube wall was rather easily determined.

$$R_t = \frac{x}{k\ A}$$

R_t = resistance to the flow of heat of tube wall, deg. F-hr/Btu

k = specific conductivity, Btu/hr-ft^2-(deg. F/in.)

x = tube thickness, in.

A = area, ft^2

The overall resistance may be determined by the following equation:

$$R = \frac{t_m}{q}$$

R = total resistance

q = heat transferred, Btu/hr

t_m = log mean temperature difference, deg. F

When air-free steam is used, the air-steam film is absent. Therefore, by subtracting the known individual resistances from the over-all resistance, the remaining resistance may be determined. This remaining resistance really consists of the water side scale, the steam side scale, and the condensate film

resistances. In this experiment, all resistances were lumped into one sum, and were assumed to remain constant as long as the condensate film temperature did not change.

When air is added to the steam, the over-all resistance is increased. The cause of this is an added resistance in the form of an air-steam film. The importance of this research work lies in the fact that this condition of a rather large percentage of air being in the steam is obtained in the lower part of a surface condenser. In other words, this eighty square feet surface condenser is made to operate as one small section of a large central station condenser.

EXPERIMENTAL RESULTS

The experimental results given in the accompanying tables and graphs are the results of tests run on an eighty square feet surface condenser located in the mechanical engineering laboratory of Kansas State College. The condenser was served by an electric motor driven centrifugal circulating pump. The steam used in this experiment came from the 100 p.s.i. College steam line.

The results shown in Table 1 are the result of tests made for the purpose of calibrating the pitot tubes. From these data, it was possible to determine the actual average velocity when the apparent maximum velocity was known. These pitot tubes were placed at the center of the condenser tubes, so that they measured the maximum velocity which occurred at the center. Fig. 5 shows the velocities throughout the condenser tubes. These curves show that the velocity of the water is zero at the tube surface and then increases to a maximum at the center. These pitot tubes were not constructed to read true velocity pressures. Actually, the pressures they read were too high, and the

velocities calculated from these pressures were called "apparent maximum velocities." For the conditions of these tests, the ratio of the actual average velocity to the apparent maximum velocity was .713.

The data of Table 2 give the manometer deflection, the apparent maximum velocity, the actual average velocity, and the total weight of water per hour for the 42 tubes in the lower pass. The manometer deflection was given in inches of nitrobenzene. The apparent maximum velocity was obtained by the following formula:

$V_m = 1.03 \sqrt{h}$

h = manometer deflection, inches

V_m = apparent maximum velocity, ft/sec

The actual velocity was obtained by multiplying the apparent maximum by .713. By knowing the cross sectional area and the average velocity, it was possible then to calculate the amount flowing. This is given in lb/hr in the last column.

Table 3, run 1, gives the data taken when there was no appreciable air at the top row. The water temperatures in and out of the various tubes are listed as well as the amount of water flowing. From these data, the amount of heat transferred in each of the tubes was easily calculated.

Run 2 gives data similar to run 1 except that air is present. Air is 15.05 percent of the mixture by weight during the run. The data were compiled and used the same as for run 1.

Runs 3 and 4 were similar in nature to runs 1 and 2 except that the air percentages were higher. During runs 3 and 4, the air was 32.9 percent and 41.1 percent of the mixture by weight.

Table 4 gives the steam and condensate temperatures. The column labeled

t_{si} gives the condensation temperature for steam at atmospheric pressure. The data of run 1 were taken on a day having a higher barometric pressure. The column t_{sbp} gives the temperatures of the steam between passes which is the same as the steam to the lower pass. The temperatures of the steam at the bottom of the lower pass and of the condensate are also given.

Certain general data are necessary, and are listed in Table 5. The mass velocities, G, are very important because they affect the heat transfer coefficients when air is present. Since the mass velocities are always raised to the .69 power for cross flow, a separate column giving $G^{.69}$ was made. The columns giving $D_{corr.}$ and $T_{corr.}$ were listed to facilitate the use of Tinker's equation.

Table 6 gives important calculated data. It was first necessary to determine the over-all coefficients for air-free steam. These, of course, were calculated from the data of run 1 during which air-free steam surrounded the tubes. The over-all resistances were the reciprocals of the over-all coefficients.

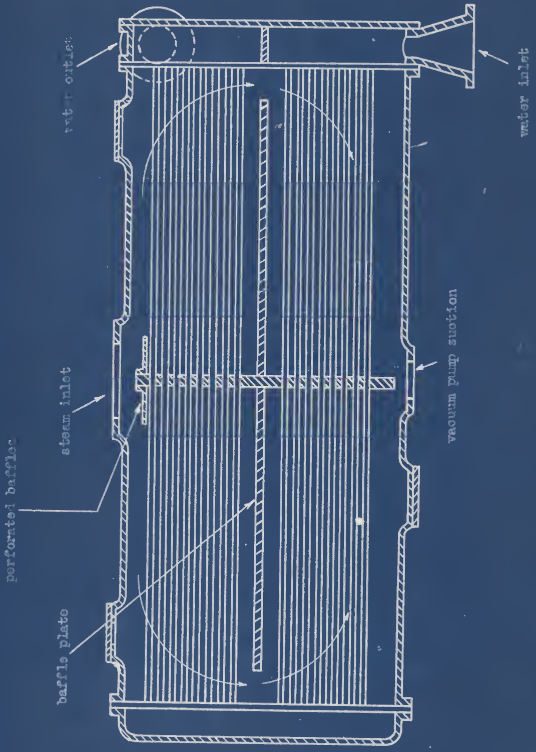
The over-all coefficients and thereby the over-all resistances for runs 2, 3, and 4 were determined. The only difference between these resistances and those calculated for run 1 is that of the added air-vapor film resistance. These air-vapor film resistances, therefore, were determined by subtracting the resistances as determined for run 1 from the resistances for runs 2, 3, and 4. The air-vapor film coefficients were the reciprocals of the corresponding resistances.

The values of "c" were found by the substitution into Tinker's equation of all known factors. This was attempted only for the top row, because the air-vapor composition was unknown for any row except the top.

EXPLANATION OF PLATE I

A drawing of the Worthington eighty square feet surface condenser.

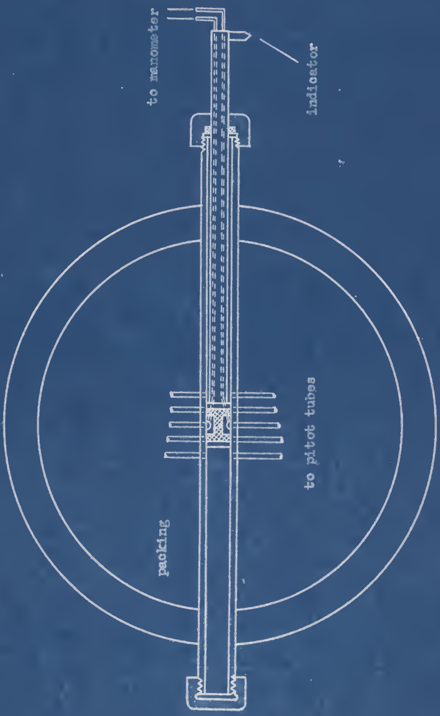
Plate I


water inlet
vacuum pump suction
steam inlet
perforated baffle
baffle plate

EXPLANATION OF PLATE II

A drawing of the pitot tube selecting device.

Plate II



EXPLANATION OF PLATE III

A drawing showing the positioning of the tubes in the condenser.

Plate III


19 15 12 8 4 21 18 14 11 7 3
21 17 14 10 6 2 20 16 12 9 5 1
20 16 11 7 3 1 19 15 10 6 2
18 13 9 5 17 13 8 4
N

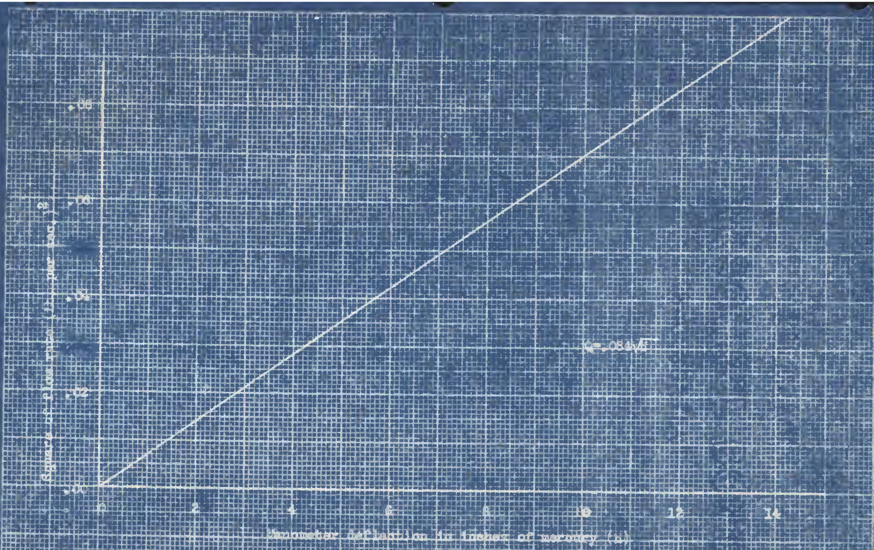


Fig. 1. Calibration of individual tube orifice.

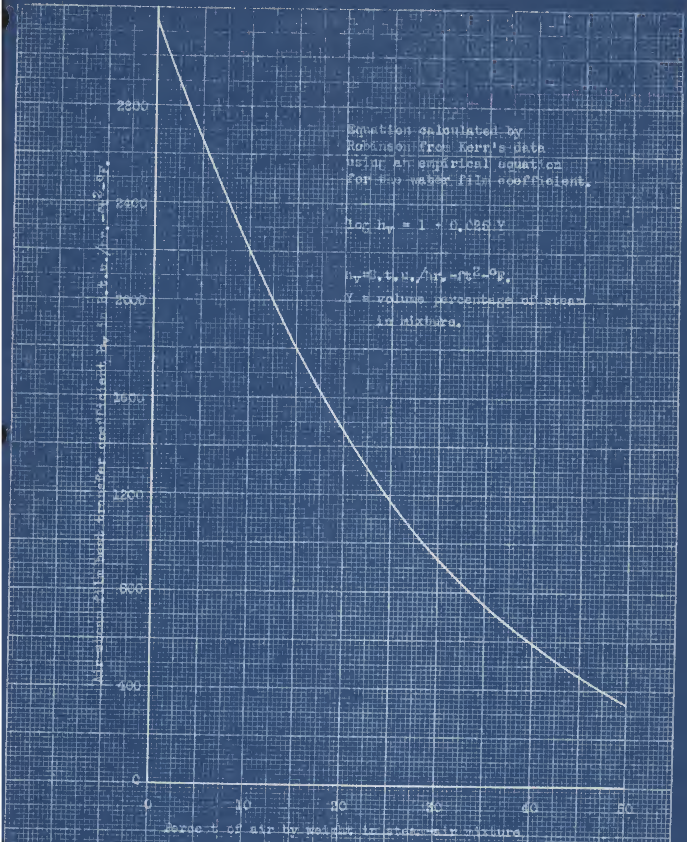


Fig. 2. Graph of Robinson's equation.

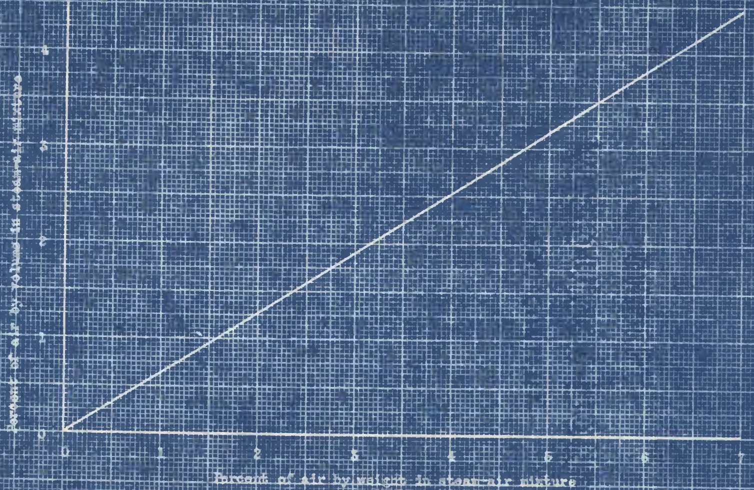


Fig. 3. Graph of weight and volume percentage relationship of steam-air mixtures for low values.

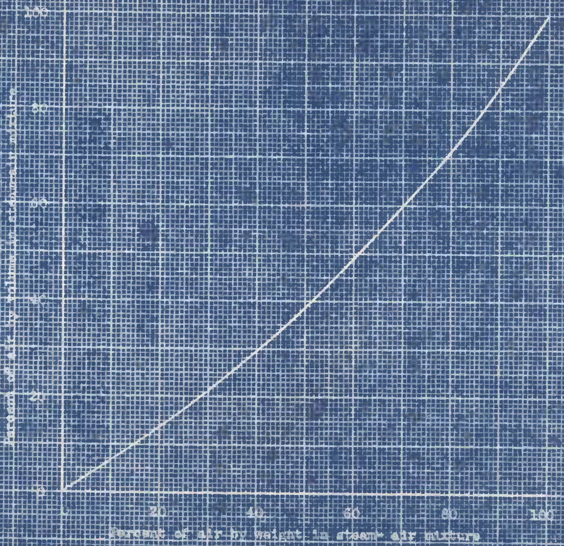


Fig.4. Graph of weight and volume percentage relationship of steam-air mixtures for high values.

Curve	Reynolds Number	Apparent Av. Vel.	Apparent Max. Vel.	Ratio of Av.Vel/Max.Vel.
A	1822	.665	1.17	.568
B	2165	.789	1.11	.711
C	3230	1.180	1.58	.747
D	4730	1.753	2.26	.776
E	6650	2.400	3.08	.780
F	7300	2.660	3.35	.793
G	10730	3.880	4.85	.800

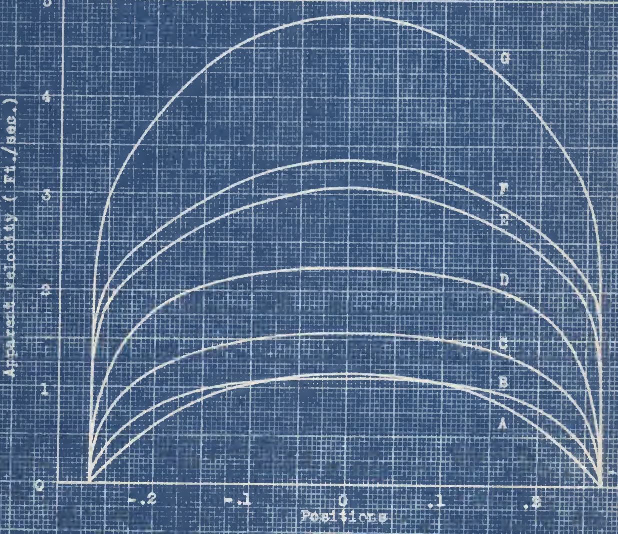


Fig.5. Individual tube velocity profiles.

Table 1. The relation of the average velocity to the apparent maximum velocity.

Tube No.	N-b. inches	Ind. Hg. man. inches	App. max. vel. ft./sec.	$V_{av.}$ ft./sec.	$\frac{V_{av.}}{V_{max.}}$
1 S	6.3	4.5	2.585	1.83	.708
1 S	8.55	6.25	3.018	2.16	.715
1 S	10.5	7.75	3.347	2.40	.717
2 S	6.8	4.65	2.685	1.863	.695
2 S	8.6	6.15	3.02	2.144	.710
2 S	10.4	7.65	3.32	2.385	.717
2 S	12.8	9.25	3.68	2.62	.712
10 N	6.7	4.65	2.665	1.864	.700
10 N	8.9	6.15	3.07	2.141	.698
10 N	12.8	9.3	3.685	2.633	.713
12 N	5.7	4.4	2.46	1.811	.737
12 N	8.0	6.15	2.91	2.145	.737
12 N	6.0	4.5	2.52	1.83	.726
12 N	9.7	7.55	3.20	2.375	.742
12 N	11.2	9.1	3.445	2.61	.757
17 N	6.45	4.65	2.62	1.861	.710
17 N	8.3	6.2	2.97	2.15	.724
17 N	10.4	7.6	3.32	2.38	.717
17 N	12.9	9.2	3.70	2.62	.708
18 N	4.4	3.0	2.16	1.496	.692
18 N	6.45	4.55	2.615	1.84	.704
18 N	9.6	6.8	3.19	2.25	.707
18 N	12.7	9.15	3.67	2.61	.712

Average constant for 7 in. Hg. total deflection is .713

Table 2. Individual tube water velocity data for 7 in. Hg. total water manometer reading.

Tube No.	Nb.	V_m	$V_{av.}$	W_w
3 N	8.20	2.95	2.100	731
7 N	6.55	2.64	1.870	652
11 N	7.32	2.79	1.990	692
14 N	7.42	2.81	2.010	700
18 N	9.45	3.17	2.260	786
21 N	9.00	3.09	3.300	765
4 S	9.90	3.24	2.310	804
8 S	7.30	2.78	1.980	689
12 S	7.32	2.79	1.990	692
15 S	6.80	2.69	1.920	668
19 S	9.55	3.18	2.270	790
1 N	7.40	2.80	2.000	696
5 N	7.20	2.76	1.970	686
9 N	7.20	2.76	1.970	686
12 N	7.30	2.78	1.980	689
16 N	6.95	2.71	1.940	675
20 N	7.28	2.78	1.980	689
2 S	7.25	2.77	1.980	689
6 S	6.35	2.60	1.860	647
10 S	6.67	2.66	1.900	661
14 S	7.33	2.79	1.990	692
17 S	7.33	2.79	1.990	692
21 S	8.30	2.97	2.120	737
2 N	6.97	2.72	1.940	675
6 N	5.87	2.49	1.780	619
10 N	5.77	2.47	1.760	613
15 N	5.98	2.52	1.800	626
19 N	5.92	2.51	1.790	623
1 S	5.62	2.44	1.740	605
3 S	5.92	2.51	1.790	623
7 S	5.35	2.38	1.700	592
11 S	6.20	2.56	1.830	637
16 S	5.42	2.40	1.710	595
20 S	7.20	2.76	1.970	686
4 N	7.25	2.77	1.980	689
8 N	6.00	2.52	1.800	626
13 N	6.15	2.55	1.820	633
17 N	6.30	2.58	1.840	640
5 S	6.02	2.53	1.810	630
9 S	5.80	2.48	1.770	616
13 S	5.77	2.47	1.760	612
18 S	6.87	2.70	1.930	672

Nb. = inches deflection of nitrobenzene manometer
V_m = apparent maximum velocity, ft/sec
$V_{av.}$ = actual average velocity, ft/sec
W_w = weight of water flowing, lb/hr

Table 3. Data showing the amount of heat transferred in each tube. Run 1

Tube No.	t_w in	t_w out	t_w	W_w	Q
3 N	81.0	131.5	50.5	731	36900
7 N	81.0	125.0	44.0	652	28700
11 N	81.0	123.0	42.0	692	29050
14 N	81.0	124.0	43.0	700	30100
18 N	81.0	125.0	44.0	786	34600
21 N	81.0	120.0	39.0	765	29850
4 S	81.0	120.0	39.0	804	31400
8 S	81.0	121.0	40.0	689	27550
12 S	81.0	123.0	42.0	692	29050
15 S	81.0	122.5	41.5	668	27700
19 S	81.0	125.0	44.0	790	34750
1 N	81.0	123.5	42.5	696	29600
5 N	81.0	121.5	40.5	686	27800
9 N	81.0	121.5	40.5	686	27800
12 N	81.0	125.0	44.0	689	30300
16 N	81.0	124.5	43.5	675	29350
20 N	81.0	120.6	39.6	689	27300
2 S	81.0	124.4	43.4	689	29900
6 S	81.0	124.4	43.4	647	28100
10 S	81.0	NG.		661	
14 S	81.0	118.0	37.0	692	25600
17 S	81.0	115.8	34.8	692	24100
21 S	81.0	127.0	46.0	737	33900
2 N	81.0	127.0	46.0	675	31000
6 N	81.0	128.5	47.5	619	29450
10 N	81.0	129.0	48.0	613	29450
15 N	81.0	129.0	48.0	626	30050
19 N	81.0	129.0	48.0	623	29900
1 S	81.0	129.0	48.0	605	29050
3 S	81.0	127.0	46.0	623	28700
7 S	81.0	129.0	48.0	592	28400
11 S	81.0	123.0	42.0	637	26700
16 S	81.0	121.0	40.0	595	23800
20 S	81.0	121.5	40.5	686	27800
4 N	81.0	120.6	39.6	689	27300
8 N	81.0	129.0	48.0	626	30050
13 N	81.0	121.5	40.5	633	25600
17 N	81.0	123.0	42.0	640	26900
5 S	81.0	120.6	39.6	630	24950
9 S	81.0	120.6	39.6	616	24400
13 S	81.0	120.4	39.4	612	24100
18 S	81.0	120.3	39.3	672	26400

Table 3. (continued) Run 2

Tube No.	t_w in	t_w out	t_w	W_w	Q
3 N	71.0	85.4	14.4	731	10530
7 N	71.0	83.6	12.6	652	8220
11 N	71.0	83.6	12.6	692	8710
14 N	71.0	82.1	11.1	700	7770
18 N	71.0	81.9	10.9	786	8560
21 N	71.0	80.5	9.5	765	7270
4 S	71.0	82.6	11.6	804	9330
8 S	71.0	81.3	10.3	689	7100
12 S	71.0	82.3	11.3	692	7820
15 S	71.0	83.1	12.1	668	8090
19 S	71.0	83.6	12.6	790	9950
1 N	71.0	80.9	9.9	696	6890
5 N	71.0	79.6	8.6	686	5900
9 N	71.0	79.1	8.1	686	5560
12 N	71.0	78.6	7.6	689	5230
16 N	71.0	78.7	7.7	675	5200
20 N	71.0	78.4	7.4	689	5100
2 S	71.0	78.9	7.9	689	5440
6 S	71.0	78.4	7.4	647	4790
10 S	71.0	NG		661	
14 S	71.0	78.1	7.1	692	4900
17 S	71.0	77.8	6.8	692	4700
21 S	71.0	81.0	10.0	737	7370
2 N	71.0	80.4	9.4	675	6340
6 N	71.0	80.4	9.4	619	5820
10 N	71.0	79.4	8.4	615	5150
15 N	71.0	80.1	9.1	626	5700
19 N	71.0	80.9	9.9	623	6170
1 S	71.0	81.7	10.7	605	6470
3 S	71.0	81.2	10.2	623	6250
7 S	71.0	80.2	9.2	592	5440
11 S	71.0	77.8	6.8	637	4330
16 S	71.0	79.0	8.0	595	4760
20 S	71.0	80.8	9.8	686	6730
4 N	71.0	78.7	7.7	689	5300
8 N	71.0	80.3	9.3	626	5820
13 N	71.0	78.9	7.9	633	5000
17 N	71.0	79.5	8.5	640	5440
5 S	71.0	80.3	9.3	630	5860
9 S	71.0	78.0	7.0	616	4310
13 S	71.0	77.4	6.4	612	3920
18 S	71.0	79.0	8.0	672	5370

Table 3. (continued) Run 3

Tube No.	t_w in	t_w out	t_w	W_w	Q
3 N	72.4	81.2	8.8	731	6430
7 N	72.4	80.0	7.6	652	4950
11 N	72.4	79.8	7.4	692	5120
14 N	72.4	79.3	6.9	700	4820
18 N	72.4	79.3	6.9	786	5420
21 N	72.4	78.2	5.8	765	4430
4 S	72.4	78.9	6.5	804	5230
8 S	72.4	78.5	6.1	689	4200
12 S	72.4	78.3	5.9	692	4080
15 S	72.4	78.6	6.2	668	4140
19 S	72.4	78.5	6.1	790	4820
1 N	72.4	77.3	4.9	696	3410
5 N	72.4	73.7	1.3	686	890
9 N	72.4	73.3	.9	686	620
12 N	72.4	72.8	.4	689	280
16 N	72.4	73.3	.9	675	610
20 N	72.4	72.7	.3	689	210
2 S	72.4	73.2	.8	689	550
6 S	72.4	73.2	.8	647	520
10 S	72.4	NG		661	
14 S	72.4	73.0	.6	692	410
17 S	72.4	72.7	.3	692	210
21 S	72.4	75.5	3.1	737	2280
2 N	72.4	77.2	4.8	675	3240
6 N	72.4	76.5	4.1	619	2540
10 N	72.4	75.7	3.3	613	2020
15 N	72.4	76.2	3.8	626	2380
19 N	72.4	76.6	4.2	623	2620
1 S	72.4	76.8	4.4	605	2660
3 S	72.4	76.8	4.4	623	2740
7 S	72.4	76.2	3.8	592	2250
11 S	72.4	75.6	3.2	637	2040
16 S	72.4	75.8	3.4	595	2020
20 S	72.4	76.4	4.0	686	2740
4 N	72.4	73.6	1.2	689	830
8 N	72.4	73.6	1.2	626	750
13 N	72.4	72.6	.2	633	130
17 N	72.4	73.0	.6	640	380
5 S	72.4	73.2	.8	630	500
9 S	72.4	72.5	.1	616	60
13 S	72.4	72.4	.0	612	0
18 S	72.4	72.8	.4	672	270

Table 3. (concluded) Run 4

Tube No.	t_w in	t_w out	Δt_w	W_w	Q
3 N	71.5	76.8	5.3	731	3870
7 N	71.5	75.9	4.4	652	2870
11 N	71.5	76.2	4.7	692	3250
14 N	71.5	76.2	4.7	700	3290
18 N	71.5	76.2	4.7	786	3700
21 N	71.5	75.5	4.0	765	3060
4 S	71.5	75.7	4.2	804	3380
8 S	71.5	75.7	4.2	689	2890
12 S	71.5	75.7	4.2	692	2910
15 S	71.5	75.4	3.9	668	2610
19 S	71.5	75.5	4.0	790	3160
1 N	71.5	73.4	1.9	696	1320
5 N	71.5	72.6	1.1	686	750
9 N	71.5	72.4	.9	686	620
12 N	71.5	72.1	.6	689	410
16 N	71.5	72.3	.8	675	540
20 N	71.5	72.3	.8	689	550
2 S	71.5	72.3	.8	689	550
6 S	71.5	72.3	.8	647	520
10 S	71.5	NG		661	
14 S	71.5	72.2	.7	692	480
17 S	71.5	72.0	.5	692	350
21 S	71.5	73.4	1.9	737	1400
2 N	71.5	74.8	3.3	676	2230
6 N	71.5	74.2	2.7	619	1670
10 N	71.5	73.8	2.3	613	1410
15 N	71.5	74.1	2.6	626	1630
19 N	71.5	74.5	3.0	623	1870
1 S	71.5	74.5	3.0	605	1815
3 S	71.5	74.5	3.0	623	1870
7 S	71.5	74.5	3.0	592	1770
11 S	71.5	74.1	2.6	637	1660
16 S	71.5	73.9	2.4	595	1430
20 S	71.5	74.5	3.0	686	2060
4 N	71.5	73.1	1.6	689	1100
8 N	71.5	72.7	1.2	626	750
13 N	71.5	72.6	1.1	635	700
17 N	71.5	72.4	.9	640	580
5 S	71.5	72.4	.9	630	570
9 S	71.5	71.8	.3	616	180
13 S	71.5	71.5	.0	612	0
18 S	71.5	71.9	.4	672	270

Table 4. Steam and condensate temperature data

Run No.	t_{si}	t_{sbp}	t_{so}	t_c
1	210.5	210.5	210.5	206.0
2	210.0	204.8	162.0	161.9
3	210.0	197.0	152.0	144.5
4	210.0	192.5	142.8	140.2

t_{si} = temperature of steam into the condenser

t_{sbp} = temperature of steam between passes of the condenser

t_{so} = temperature of steam at the bottom of the condenser

t_c = temperature of the condensate from the hotwell

Table 5. General data

Run No.	G	$G^{.69}$	$D_{corr.}$	$T_{corr.}$
1			1.157	
2	64.50	17.70	1.157	.735
3	19.35	7.70	1.157	.750
4	17.13	7.10	1.157	.759

Table 6. Important calculated data

Tube No.	t_m	U	R	R_o	h_a	t_t	P_{gf}	P_{vf}	P_{vf}/P_{gf}	f	c
					Run 1						
3 N	102.0	371	.00270	.00270							
7 N	106.0	278	.00360	.00360							
11 N	107.0	278	.00360	.00360							
14 N	106.5	290	.00345	.00345							
18 N	106.0	334	.00299	.00299							
21 N	109.0	281	.00356	.00356							
4 S	109.0	295	.00339	.00339							
8 S	108.2	261	.00383	.00383							
12 S	107.0	278	.00360	.00360							
15 S	107.5	264	.00379	.00379							
19 S	106.0	336	.00298	.00298							
		296.9 av U									
					Run 2						
3 N	126.3	85.5	.01170	.00270	111.1	110.0	10.53	18.23	1.730	.837	5.10
7 N	127.2	66.3	.01510	.00360	86.9	110.3	10.52	18.24	1.732	.837	3.98
11 N	127.2	70.2	.01423	.00360	94.0	112.2	10.48	18.28	1.742	.838	4.27
14 N	127.9	62.3	.01604	.00345	79.5	107.8	10.56	18.20	1.724	.837	3.66
18 N	128.0	68.6	.01458	.00299	86.2	105.6	10.61	18.15	1.710	.836	4.00
21 N	128.7	58.0	.01726	.00356	73.0	105.3	10.62	18.14	1.709	.836	3.39
4 S	127.7	74.9	.01335	.00339	100.4	111.8	10.49	18.27	1.741	.838	4.58
8 S	128.3	56.7	.01760	.00383	72.7	107.1	10.57	18.19	1.722	.837	3.35
12 S	127.8	62.7	.01595	.00360	81.0	108.3	10.55	18.21	1.726	.837	3.72
15 S	127.4	65.1	.01536	.00379	86.5	111.2	10.50	18.26	1.736	.837	3.94
19 S	127.2	80.2	.01248	.00298	105.3	110.3	10.52	18.24	1.732	.837	4.82
		69.59								av.	4.07

Table 6. (concluded)

Run 3

3 N	120.0	55.0	.01820	.00270	64.5	97.3	14.58	14.18	.972	.775	12.82
7 N	120.6	42.1	.02380	.00360	49.5	97.0	14.59	14.17	.971	.775	9.85
11 N	120.7	43.5	.02300	.00360	51.5	97.7	14.57	14.19	.973	.775	10.22
14 N	121.0	40.8	.02450	.00345	47.5	95.5	14.61	14.15	.968	.775	9.48
18 N	121.0	45.9	.02180	.00299	53.2	95.0	14.62	14.14	.967	.775	10.63
21 N	121.5	37.4	.02670	.00356	43.2	94.5	14.63	14.13	.965	.774	8.66
4 S	121.2	44.3	.02260	.00339	52.0	96.5	14.59	14.17	.971	.775	10.35
8 S	121.4	35.5	.02820	.00383	41.0	94.7	14.62	14.14	.967	.775	8.20
12 S	121.5	34.5	.02900	.00360	39.4	93.2	14.65	14.11	.963	.774	7.92
15 S	121.3	35.0	.02850	.00379	40.5	94.6	14.63	14.13	.965	.774	8.12
19 S	121.4	40.7	.02460	.00298	46.2	93.0	14.65	14.11	.963	.774	9.28
		41.34 av v									av. 9.59

Run 4

3 N	118.3	33.6	.02980	.00270	36.9	87.7	16.32	12.44	.762	.748	10.40
7 N	118.7	24.8	.04030	.00360	27.2	87.2	16.33	12.43	.761	.748	6.13
11 N	118.6	28.1	.03560	.00360	31.2	88.5	16.30	12.46	.764	.748	8.77
14 N	118.6	28.5	.03510	.00345	31.6	88.3	16.30	12.46	.764	.748	8.87
18 N	118.6	32.0	.03130	.00299	35.3	87.8	16.31	12.45	.763	.748	9.93
21 N	118.9	26.4	.03790	.00356	29.1	87.4	16.32	12.44	.762	.748	8.18
4 S	118.8	29.2	.03430	.00339	32.4	88.1	16.31	12.45	.763	.748	9.11
8 S	118.8	25.0	.04000	.00383	27.7	88.0	16.31	12.45	.763	.748	7.79
12 S	118.8	25.1	.03980	.00360	27.6	87.2	16.33	12.43	.761	.748	7.77
15 S	118.9	22.5	.04450	.00379	24.6	86.3	16.34	12.42	.760	.748	6.94
19 S	118.9	27.3	.03670	.00298	29.7	86.0	16.35	12.41	.760	.748	8.38
		27.9									av. 8.39

DISCUSSION

There are three controllable factors governing heat transfer in a surface condenser. These factors are (1) the water velocity, (2) the steam temperature, and (3) the mixture mass velocity. All of these factors are influenced by the design of the condenser.

The water distribution between the many tubes in a steam condenser is determined by the design of the water boxes. The water boxes are those chambers into which the water is sent before and after it has passed through the condenser tubes.

The steam temperature is partially governed by the pressure loss occurring as the steam flows through the condenser. This, of course, is determined by the tube configuration in the condenser shell. Sufficient velocity should be attained by the steam-air mixture so that a reasonably high coefficient is realized, but not to such an extent that the pressure loss is excessive. The mass velocity may be changed by increasing or decreasing the free area through which the steam must pass.

The data indicate that there was quite a variation in water velocity between the various tubes. However, a somewhat general pattern was found. Table 2 indicates that except for 2 tubes in the second row, all tubes having a velocity of 2 ft/sec or more are in the top row. The average amounts of tube water flow for the four rows, starting at the top, were 724, 686, 627, and 639 lb/hr. The water baffle, against which the incoming water was directed, changed the velocity head into pressure head. This occurred near the top row of tubes, and probably accounts for the greater flow in this row. The two

tubes in the second row having the high velocities were on the ends of this row. Except for the top row, there is a marked tendency for the end tubes of a row to have higher velocities than the other tubes in the row. For the 42 tubes of the lower pass, the maximum and the minimum velocities were 2.26 and 1.70 ft/sec respectively. This would indicate that there is a variation from 12.9 percent lower to 15.8 percent higher than average. The water velocity is a more important factor in the main condensing section of a condenser than in the air cooling section, because the water film resistance is a larger percentage of the total resistance.

As the amount of water sent through the condenser is increased, the variations of individual tube water velocities are increased. For the average velocity used in this experiment, the variations of individual tube water velocities are not particularly important. However, commercial condensers usually have water velocities of 4 ft/sec or more instead of 2 ft/sec, and so the variations may become very important.

The over-all coefficient of one of the tubes of the upper row (3 N) was 371 Btu/hr-^{o}F-ft^2 when it was surrounded by pure steam. Throughout the tests, this tube had the highest coefficient. Apparently, there were two factors tending to make the amount of heat transferred in this tube rather high. These factors were (1) a higher than average water velocity, and (2) a high steam velocity toward this section of the condenser. There was an elbow in the steam inlet pipe just about a foot above the condenser. This tended to give the steam an initial velocity toward the north half of the condenser.

One thermocouple, number 10 S, was bad. This was probably due to a short, and it was not at all surprising because these thermocouples were not easily installed. It was fortunate that only one thermocouple out of the 42

was bad.

When pure steam was used, all tubes tended to transfer more nearly the same amount of heat. Certain tubes did not transfer as much heat as they should. These tubes were located in the second and fourth rows. Also, there was one tube in the third row that transferred a comparatively low amount of heat. There were four rows of tubes in the lower pass of the condenser. Starting from the top, the number of tubes in these four rows were 11, 12, 11, and 8. The center to center distance between tubes measured 15/16 inch which gave a clearance between adjacent tubes of 5/16 inch.

As the air percentage increased, the tubes in the second and fourth rows rapidly became less useful. This should not happen in the second row. The fact that all the tubes in the second row, except the end tubes, became less efficient seemed to indicate that these tubes were not placed as well as they might be. The data indicate that in nearly every case, the end tubes transferred much more heat than the inside tubes. This difficulty became apparent when air was 15.05 percent of the mixture by weight. However, it became very serious when the air reached 32.9 percent. It would seem that for pure vapor condensation, the tube positioning is satisfactory, although it could be improved. However, for the air cooling section of a condenser, the tube positioning is definitely bad. In all probability, this condenser was designed by a draftsman, and was thereby made with a geometric pattern regardless of its performance.

The value of "c" was determined from the coefficient of the air-vapor film, h_a. Also, it was necessary to determine the relative thickness factor f.

$$f = \frac{c2 + Z}{c_3 + Z}$$

c_2 and c_3 = constants

$Z = P_{vf}/P_{gf}$

It was assumed that c_2 was .556 and that c_3 was 1.0. The term P_{gf} was the logarithmic mean pressure of the air in this film. The pressure of the vapor was found by subtracting P_{gf} from the total pressure in the condenser. These values were calculated for the top row only, since the steam temperatures were known only at this row.

The variations in the tabular values of "c" for the various tubes do not necessarily represent the results of experimental errors. They are mainly caused by assuming an average value of mass flow (G) instead of using the values of mass flow for each of the many tubes. It was impossible to determine these individual values of mass flow. This fact might explain why the value of "c" given by Tinker did not agree with these data. However, "c" as determined by an experiment on a condenser having a number of tubes will probably be more valuable to the design engineer than one calculated from data on a single tube.

The value of "c" in Tinker's equation does not appear to be a constant. The data seem to indicate that the value of "c" varies as the composition of the mixture changes, or in other words, with the ratio w_a/w_v. A great deal more experimental work would be necessary to definitely determine the variation of "c".

The effect of a small percentage of air is surprising. When air comprised only 15.05 percent of the mixture by weight, the average over-all coefficient was reduced to 23 percent of the air-free steam coefficient. The individual tube over-all coefficients, as found in the top row, varied from 20.5 percent to 25.4 percent of the air-free steam coefficients. It may

readily be seen, that the design of the air cooler section is extremely important. Even with the best possible design, the air cooler section requires a rather large amount of surface. With a poor design, the situation is worse. Of course, the problem cannot be solved in the best possible manner unless all equipment is considered. This particularly applies to the air pump. An increase in mixture velocity increases the heat transfer coefficient, but also increases friction drop in the air cooler. This would call for a larger air pump, which would be more expensive to operate. These factors must be balanced to the best economic design, and reliable data of the type found here must be available.

CONCLUSIONS

For the operating conditions used, the tubes in the upper row were too close together. When a large percentage of air is present, the second row could be removed without materially changing the capacity of the condenser. Perhaps with different operating conditions, the situation might be different.

The water distribution could be improved by changing the water box design. The water box might be made with a tapering end. The manufacturer may feel that the added cost for the changed design would not be justified.

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ABSTRACT OF THESIS

HEAT TRANSFER IN AN EIGHTY SQUARE FEET SURFACE CONDENSER

by

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PURPOSE

The purpose of this research work is that of studying the distribution of heat transferred in a multitube condenser. The common practice in studying heat transfer is that of constructing a single tube condenser. The construction of a single tube condenser is a very simple method of attack, but it is doubted that data thus found can be applied to a multitube condenser, without modifications. At least, some examination of the process of heat transfer in a multitube condenser should be made because multitube condensers are always used in steam power plants, and never single tube condensers.

GENERAL METHOD

The equipment used for this work was a Worthington eighty square feet surface condenser. This condenser had eighty two tubes, but only forty two tubes were used in this investigation. The tubes in this condenser were 5/8 inch outside diameter, and were of 18 B.w.g. wall thickness.

First, it was necessary to determine the amount of water to each tube. These data were obtained by installing a pitot tube in each of the forty two tubes. A device to connect each of these pitot tubes, in order, to a manometer fastened to the outside of the condenser, was designed. By means of these pitot tubes and a preliminary calibration, the weight of water through each of the tubes was determined.

The temperature of the water entering the condenser was

uniform, and easily determined. This temperature was obtained by a thermometer, and checked by a thermocouple.

The water temperatures from the forty two tubes were determined by thermocouples placed at the outlets of these tubes. There was no metal contact between these thermocouples and the condenser. This makes certain that the thermocouples read the water temperature and not the metal temperature. An external switchboard connected the thermocouples to a potentiometer.

The tests were run at atmospheric pressure. It was ~~The writer~~ intended to operate the condenser under a 28 inch vacuum, but the condensate pump refused to work when a vacuum was obtained in the condenser. Since the pressure throughout the condenser was atmospheric, it was a simple matter to get the air pressure at any point in the condenser if the vapor pressure at this point was known. The vapor pressure was determined by obtaining the temperature of the vapor. When the vapor is saturated, the temperature and pressure correspond.

The temperatures of the steam-air mixture between passes and after the last pass were obtained. The steam-air mixture arriving at the top row of tubes in the lower pass was at the temperature between passes.

The total flow past this top row was determined by obtaining the total heat transferred in the tubes below it, and dividing this by the latent heat per pound. To calculate the

mass flow, it was only necessary to divide the total flow per hour by the net minimum cross sectional area.

The condenser was operated with no air at the top row of condenser tubes, and three other tests were run with three different amounts of air. From the data taken in the test when no air was present, the overall coefficients of heat transfer for air free steam were found. The reciprocal of this coefficient is the sum of the resistances of the water, scale, tube, and condensate films. The sum of these resistances was considered constant, although actually this ~~is~~ was not quite true. However, the steam film resistance decreases while the water film resistance increases, and they will somewhat balance each other.

The coefficients of heat transfer for the other tests were determined. The resistances were determined by taking the reciprocals of these coefficients. The only difference between these resistances and the previous values was the added resistance of the air-vapor film. Therefore, it was possible to get the value of the air-vapor coefficient.

These values of the air-vapor coefficients were used to obtain values of "c" in the Tinker equation. This equation is given in such form that it may readily be used in practical problems.

RESULTS

The most important calculated factor is probably the overall coefficient of heat transfer. This coefficient is determined by dividing the heat transferred per hour per square foot by the mean temperature difference between the water and the steam. This mean temperature difference is usually taken as the log mean difference.

$$t_m = \frac{\theta_1 - \theta_2}{\log_e \frac{\theta_1}{\theta_2}}$$

t_m = log mean temperature difference

$\theta_1 = t_s - t_{win}$

$\theta_2 = t_s - t_{wout}$

t_s = temperature of the steam

t_{win} = temperature of the water in

t_{wout} = temperature of the water out

The overall resistance to the flow of heat is the reciprocal of the overall coefficient. This overall resistance consists of several smaller resistances as previously stated.

Four runs were made starting with zero air for run 1, and using increasing air percentages until 41.1% was obtained during run 4.

Run 1, using air-free steam, showed that the overall coefficients for the top row ranged from 261 to 371 Btu/hr-°F-ft^2. An explanation for this variation is not easy to find. The

water velocity in the tube having the 371 coefficient was higher, and the water film had 10% less resistance to the flow of heat. Also, there may have been some difference in the amount of scale on these tubes. This scale can not be easily measured.

The coefficients found in run 1 were assumed to be the same for the remaining test runs except for the addition of the air-vapor film. This air-vapor film coefficient ranged from 72.7 to 111.1 Btu/hr-°F-ft^2 for the various tubes during run 2. The average mass flow velocity for the top row during this run was 64.5 lb/hr-ft^2. The air in this run was 15.05% of the mixture by weight. The data indicate that for these operating conditions, the value of "c" in Tinker's equation should be 4.07. Tinker's equation follows:

$$h_a = c\,(T_{corr.})(D_{corr.})\,G^{.69}f\,\frac{P_{vf}}{P_{gf}}$$

h_a = air-vapor film coefficient

c = a constant

$T_{corr.} = \left(\frac{560}{T}\right)^{1.8}$

T = degrees absoulute (Rankine)

$D_{corr.} = \left(\frac{1}{d}\right)^{.31}$

d = tube diameter, inches

G = mass velocity, lb/hr-ft^2

P_{vf} = mean vapor pressure, in. Hg

P_{gf} = log mean gas pressure, in. Hg

f = film thickness factor depending on relative concentrations of gas and vapor or on $\frac{P_{vf}}{P_{gf}}$

$$P_{gf} = \frac{P_{gt} - P_{gm}}{\log_e \frac{P_{gt}}{P_{gm}}}$$

$P_{vf} = P - P_{gf}$

$P_{gt} = P$ - (water vapor pressure corresponding to T_t)

$P_{gm} = P$ - (water vapor pressure corresponding to T_m)

T_t = tube wall temperature, °F

T_m = air-vapor mixture temperature, °F

P = total mixture pressure, in. Hg

For run 3, the air-vapor film coefficients varied from 39.4 to 64.5 Btu/hr-°F-ft^2. The average mass flow was 19.35 lb/hr-ft^2. The air was 32.9% of the mixture by weight. The average value of "c" for this run was 9.59.

The air-vapor film coefficient ranged from 24.6 to 36.9 Btu/hr-°F-ft^2 for run 4. The mass velocity for this run was 17.13 lb/hr-ft^2. The air was 41.1% of the mixture by weight. Using the above values in Tinker's equation, an average "c" of 8.39 was obtained.

From the above data, it would seem that "c" is not a constant. However, these tests were run over a somewhat wider range of air percentages than would ordinarily be encountered in steam condensers, and possibly over a limited range, "c"

may be assumed constant.

In this work, however, the condensate temperature was used instead of the tube wall temperature as recommended by Tinker. It was felt that a film coefficient ought to be given in terms of the boundary conditions of the film itself. This makes the "c" thus obtained just a little lower than Tinker's value.

The data indicate that when there was plenty of air-free steam, all tubes condensed a reasonable amount of steam. However, as the percentage of air in the mixture was increased, certain tubes began to transfer a smaller percentage of the heat transferred. These tubes were in the second and fourth rows. There were four rows of tubes in the lower pass of the condenser. Starting from the top, the number of tubes in these four rows were 11, 12, 11, and 8. The center to center distance between tubes measured 15/16 inch which gave a clearance between adjacent tubes of 5/16 inch.

All tubes in the second row were shielded by other condenser tubes except for the two tubes on the ends of this row. The last or fourth row was particularly weak in runs 3 and 4. The reason for this is, that by the time the steam reaches the bottom row, so much of the steam has been condensed that the air percentage is very high. The percentages of air previously given apply only at the top row, and at the bottom row they will be much higher. The low coefficients of the bottom row were not surprising, but the low coefficients of the second

row were not expected. It appears that the tubes were placed by a draftsman and not by a design engineer. More steam should be sent to the second row.

The top row tubes passed the highest average amount of water which was 724 lb/hr. The second, fourth, and third follow with amounts of 686, 639, and 627 lb/hr respectively. Some local internal condition probably accounted for the higher velocity in the fourth row. The water baffle, against which the incoming water was directed, changed the velocity head into pressure head. This occurred near the top row of tubes, and probably accounts for the greater flow in this row.

The data seemed rather consistant, and somewhat symetrical about a vertical center line. This was especially true of the water velocities, but it was also true to a considerable extent of the heat transfer data. It is felt that the data given in this thesis may be relied upon for the conditions of operation used.